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The Simulation and Design of a High Efficiency, Lubricant Free, Linear Compressor for a Domestic Refrigerator

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THE SIMULATION AND DESIGN OF A HIGH EFFICIENCY, LUBRICANT FREE, LINEAR COMPRESSOR FOR A DOMESTIC REFRIGERATOR.

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ABSTRACT

The first phase of a project at Sunpower, Inc. sponsored by the EPA is described. The program involves the design, computer simulation and fabrication of a high efficiency linear compressor for application in domestic refrigeration. The use of a high efficiency electric motor and non-contact operation can dramatically reduce the electrical and friction losses associated with crank compressors. This technology has already been proven at Sunpower in prototypes of Stirling cycle refrigerators. The computer simulation predicts stable operation of the design under all conceived operating conditions and indicates that a COP of 2.0 can be achieved for a heat lift of 250 Watts from -23 C to 32 C. Early tests conducted on the prototype support these findings.

INTRODUCTION

Rotary compressors for vapor compression cycles are highly developed but still have significant losses associated with them. The free piston linear compressor has the potential to reduce these losses substantially and achieve better efficiency. It also promises life and cost benefits and has the added advantage of being able to operate without lubrication or wear.

This paper describes the first phase of a project at Sunpower, Inc. sponsored by the USA Enviromental Protection Agency. The program involves the design and computer simulation of a high efficiency linear compressor for application in a domestic refrigerator. At the completion of the project, a functional prototype will be tested by an independent third party. The unit is designed for operation with R12 refrigerant so that direct comparison with existing units can be made. However, with minor modifications it could be used with any similar refrigerant such as R22 or HFC 134a. It could also be used for compressing any chemically compatible gas and, since it requires no lubrication, will not contaminate such a gas. This paper outlines a computer simulation used to study the dynamics of the system and presents preliminary test results.

Introduction to Free Piston Devices

In a free piston compressor the piston is not rigidly attached to a driving mechanism such as a crank. Because of this its motion is not constrained by the geometry of the driver. Both the stroke and mean position of the piston can change and these are dictated by the mechanical, magnetic and pressure forces acting on it. This can be a disadvantage since the piston motion is not predefined making it necessary to have some mechanism to control its position, particularly when fragile parts might collide. However, it makes the machine more versatile since the piston motion can be adjusted continually to achieve optimum performance. Another advantage of a free piston device is that because all the driving forces act along the line of motion, there is no sideways thrust on the piston. This reduces the high friction losses and wear rates associated with crank devices.

A compact electromagnetic mechanism can't generate the high forces needed to drive the piston of a compressor. To approximate the sinusoidal piston motion desired the piston is sprung so that its natural frequency is close to the intended operating frequency. This springing can be provided mechanically, electrically or by pressure forces or any combination of the three. By driving the system close to resonance, a large amount of energy can be transferred to the system without the need for a large driving force. However, if the driving frequency or natural frequency change, the system will move away from resonance and its power capacity will decrease. Thus the tuning of the system has a significant effect on its response.

IMPROVEMENTS OVER EXISTING COMPRESSORS

Efficiency

An Americold model SG-108, representative of the current state of the art in compressors for domestic refrigeration, was chosen as the basis of comparison for the compressor. This unit is a conventional crank unit with a 25.4mm diameter piston driven by a 60Hz AC induction motor. It draws 162W of electrical power when lifting 247W of heat from -23 C to 32 C. The losses in this unit at this operating point can be broken down as follows:

Loss in electrical motor:	26W
Friction/ viscous shear on piston:	24W
Other Losses	25W

The "other losses", which weren't separated, consist of the gas hysteresis in the compression space; the flow losses due to a pressure drop across the valves and leakage past the piston and through the valves.

The linear compressor uses the same valve head and has a similar bore, stroke and clearance volume to the Americold compressor. Thus one would expect the valve and hysteresis losses to be similar. However, the piston seal on the linear compressor is significantly different - it is longer, has more clearance and does not have the benefit of the sealing action of the lubricating oil. This makes it difficult to predict the relative loss due to leakage past this seal.

Sunpower has routinely produced electric motors with efficiencies above 90 percent. An efficiency of 94 percent is expected at 150W input power for the designed motor. This would yield a saving of 16W input power.

Non contact operation of the piston makes it virtually friction free since the loss due to gaseous shear is negligible and there is no metal to metal contact. This has been verified at Sunpower in experiments conducted on non contact Stirling machines, where no power loss due to friction on the piston could be detected. A net saving of 23W is expected from the elimination of piston friction and viscous shear.

Thus, if the leakage past the piston of the linear compressor is similar to that in the Americold compressor, the power required to lift 247W of heat from -23 C to 32 C should be approximately 122W. This would yield a COP of 2.0, compared to 1.5 for the crank compressor. This is a 33% improvement in efficiency!

Other Improvements

Additional benefits of the linear compressor are that, due to its non-contact, oil free operation, there will be no contamination of the working fluid. Thus the effectiveness of the heat exchangers should be improved. A long, trouble free life can be expected. Also, by removing the need for a lubricant, a major hurdle to the use of the ozone friendly refrigerant HFC 134a is removed.

Since the stroke of the machine can be controlled by adjusting the input voltage, power modulation should be feasible. This would remove the losses and high startup currents associated with on/off control.

Cost

The cost of the unit should be similar, if not less than that for a conventional compressor. There is one moving part, compared to three for a crank machine. Consequently there is only one bearing surface which requires a single close clearance fit similar to those in a crank machine. The motor uses less copper and iron than the Americold unit.

COMPUTER SIMULATION

Description of Model

The compressor was modelled using ACSL, a program which uses a time-stepping method to give a numerical solution to a set of differential equations. The primary purpose of simulating the compressor was to investigate the dynamics of the device, particularly its stability. The model was also used as a tool to determine the values of physical parameters needed to tune the system and to give a rough idea of the performance that might be achieved in practice. The model has three basic parts: a permanent electric motor; the thermodynamics of the compression space and the forces acting on the piston.

The electric motor was modelled as a series circuit containing the following components: an applied sinusoidal voltage of constant frequency and amplitude; an inductor; a capacitor; a resistor and a generated voltage. The applied voltage represents the power supply to the motor. The inductor represents the coil. The capacitor represents an external capacitor used to tune the electrical circuit for 60Hz. The resistor represents the sum of the iron loss, magnet loss, eddy current loss, magnetic fringing loss and the DC resistance of the coil. The generated voltage is the back EMF induced in the circuit by the motion of the magnets. The differential equation for the electric circuit is thus:

$$\frac{dI}{dt} = \frac{1}{L}\left(V \sin(\varpi t) - \alpha \ddot{x} - \frac{1}{C}\int I\, dt - I R_{AC} \right)$$

where:

I = Current through coil
$\ddot{x}$ = Piston accelleration
t = Time
α = Constant linking electromotive force and current for motor
V = Amplitude of supply voltage
ϖ = Frequency of supply voltage
L = Inductance of coil
C = Capacitance
R_{AC} = AC resistance of coil

The thermodynamic pumping cycle was modelled by isentropic compression and expansion processes connected by constant pressure intake and discharge processes. Pressure drops of 0.15 bar and 0.4 bar were assumed across the intake and discharge valves respectively. The density of the incoming charge was assumed to be that of R12 at the inlet pressure and ambient temperature. The isentropic index of the working fluid was assumed to have a constant value of 1.16. The model did not attempt to account for gas hysteresis or leakage of gas past the valves and piston seal. The pressure acting on the back of the piston was assumed to be constant.

Newton's second law was applied to the piston to obtain an equation for its acceleration. The forces acting on the piston are: the pressure forces on the front and back surfaces; the force created by the deflection of the spring (which was assumed to be linear) and the magnetic force applied by the electric motor. Windage forces were neglected. The effect of casing motion was neglected since the mass of the casing is much larger than that of the piston assembly. The equation used was thus:

$$\ddot{x} = \frac{1}{m}\left(\alpha I - A_p (P_w - P_b) - K \right)$$

where:

m = Mass of piston
A_p = Area of piston face
P_w = Pressure in compression space
P_b = Pressure in back end
K = Spring stiffness

The vapor compression cycle was modelled by assuming a constant inlet pressure dictated by the vapor pressure of R12 at the source temperature. The discharge pressure was assumed to be the vapor pressure of R12 at the temperature of the condensor. This temperature is a function of the mass flowrate, the ambient temperature and the effectiveness of the heat exchanger.

Non-Linearities in System Response

The non-linearities of the system arise primarily from the constant pressure intake and discharge processes. The effect of these on the current and piston acceleration are apparent in Fig 1 and Fig 2. Two basic configurations were compared, one with a relatively light reciprocating mass and little springing and one with a heavier mass which required more springing to resonate. The acceleration and current are closer to sinusoidal for the system with the heavy piston. Higher harmonics in the current are undesirable since they create resistance losses but cause no net useful work to be done on the gas. This results in a decrease in the efficiency of the motor which becomes significant at low input powers. The higher harmonics in the current were more apparent at low input powers, particularly the 120Hz component which created a second peak in the waveform for the light piston system.

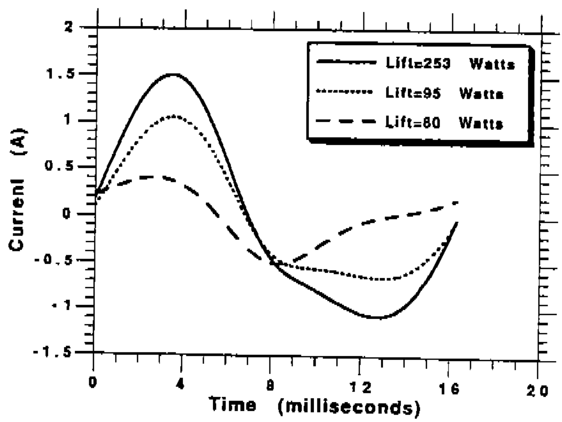

Fig 1: Motor Current at 250 Watts Heat Lift.

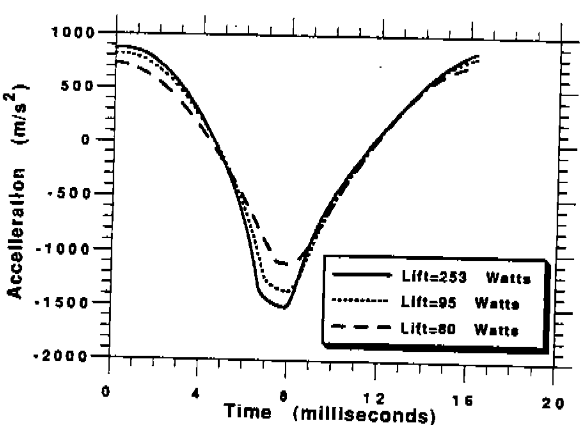

Fig 2: Piston Acceleration at 250 Watts Heat Lift.

Higher harmonics in the piston acceleration are undesirable since they make it more difficult to balance the compressor using a dynamic absorber. However, if the vibration of an unbalanced unit mounted on light springs is within acceptable limits this may be irrelevant. The system with the light piston also detunes more rapidly as the input voltage is reduced to simulate power modulation.

Stability

The only configuration modelled where the system was not stable was when the piston was oversprung. Here the piston amplitude oscillated causing the piston to repeatedly strike the head. However, if suitable values of piston mass and spring stiffness were chosen the system response would stabilize within a few cycles after a change in the input voltage of back end pressure. When started from rest, the system would reach steady state within sixty cycles. Of course in practice it would take much longer to reach steady state because of the thermal inertia of the heat exchangers.

Control of Piston Position

To maintain a high volumetric efficiency it is desirable to have the piston run as close to the head as possible to minimize the dead volume. Doing this will maximize the capacity of a compressor of given stroke and piston area. On a crank machine the piston motion is defined by the geometry of the machine. However, on a free piston device the possibility exists that the piston could move in from its mean position and strike the head or could move out and degrade the performance of the machine. The simulation showed that if the back end pressure was equal to the intake pressure, the piston mean position would drift outwards (away from the head) as soon as pumping work was done, resulting in an increase in the dead volume giving a very low pumping capacity. This is caused by an increase in the mean pressure in the working space corresponding to an increase in pumping which pushes the piston out. To prevent this from happening an opposing force is required to push the piston inwards.

Three ways to achieve this were considered. The first is to bias the piston inward slightly. It was found that to hold the piston at its mid position extremely stiff springs are needed. The second possibility is to superimpose a DC component on the driving voltage which results in a net inward force on the piston. However, it increases the magnetic flux load on the motor laminations and creates an electric loss in the resistance of the coil. The third possibility is to adjust the pressure on the back of the piston to control its mean position.

In the simulation, the piston was maintained at its mid position at 250W of lift. As the input voltage is reduced the piston drifts in slightly towards the head and the amplitude decreases as shown in Fig 3. The stiffer the springing on the piston, the less this drift was. The data shown is for a piston mass of 700g and a spring stiffness of 70 N/mm.

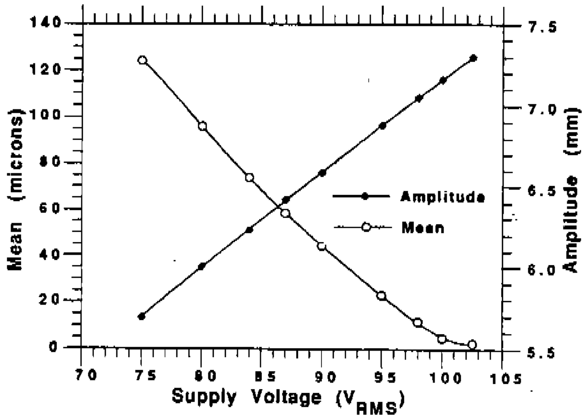

Fig 3: Effect of Supply Voltage on Piston Motion.

Performance and Modulation

The simulation indicates that a lift of 253W from a source at -23 C to an ambient temperature of 32 C should be achieved at a piston amplitude of 7.3 mm. A COP of 2.8 is predicted at this lift. However, this is optimistic since the only loss modelled is the electrical resistance loss in the motor. If an additional loss of 27W is factored in at all lifts to account for leakage, flow and gas hysteresis losses the results of Fig 4 and Fig 5 are generated. This is conservative since at lower discharge rates these losses should decrease. These performance curves are what can reasonably be expected from the hardware.

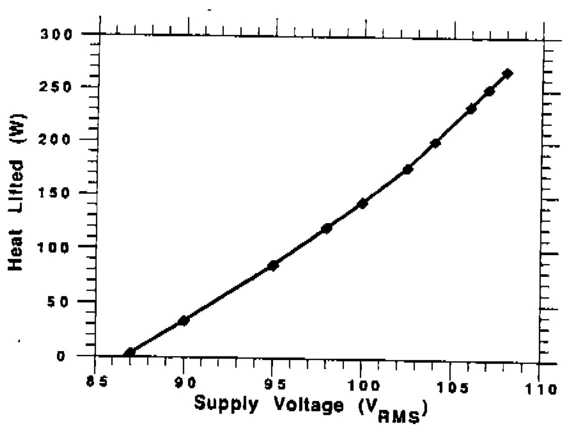

Fig 4: Modulation of Lift

As can be seen, the heat lifted is almost a linear function of the input voltage, which should make it very easy to control the lift to match the load requirement. The COP is impressively high over a wide range of lifts. It is above 2.0 for all lifts above 180 Watts and only drops below the current standard of 1.5 when the lift drops to 80 Watts. The reduction in COP with lift is a result of the losses becoming a larger proportion of the input power as the lift is reduced.

HARDWARE

Description of Test Rig

The linear compressor was installed in series with a condensor and evaporator taken from a domestic refrigerator. The test rig was designed to simulate operation in a domestic refrigerator. The load on the evaporator is controlled and measured via electric surface heaters attached to two aluminum plates which sandwich the evaporator. The condensor is mounted in an enclosure which allows control of its temperature via restriction of the cooling air flow.

Thermocouples are distributed through the system to measure temperatures at various points in the cycle. The pressure at the intake of the condensor, the exit of the evaporator and in the back end of the compressor are monitored with precision test gauges. The piston position and the compression space pressure were measured dynamically with an FLDT and a miniature strain gauge pressure transducer respectively. The voltage, current and power driving the compressor were measured. The flow of vapor into the compressor was measured with a variable area flowmeter.

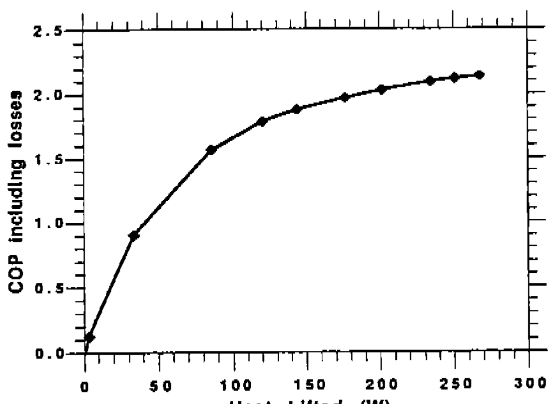

Fig 5: Predicted Performance

Inventory of Losses

Before installation into the test rig various tests were conducted on the compressor to estimate losses. The AC resistance was measured to be very close to what was expected indicating that a motor efficiency of 94.5 percent at 162W input power should be achieved. This relates to a saving of 16W over a commercial AC inductance motor of similar size. The motor contains 230g of permanent magnets, 900g of copper and 1430g of iron. This design could be shrunk to give a motor of 91.8 percent efficiency at 162W containing approximately 170g of magnets, 360g of copper and 1120g of iron.

The loss due to piston seal leakage was measured to be approximately 4W. Valve leakage under static conditions was very small.

Preliminary Results

For debugging purposes the unit was used to pump air from ambient pressure to 13.8 bar, which is similar to what it would experience in a refrigerator. During these runs the unit exhibited all the behavior trends predicted by the simulation. Its pumping efficiency was measured to be approximately 65 percent when delivering 60W of pumping power. This corresponds to an efficiency of 75 efficiency when R12 is pumped since gas hysteresis losses for that medium should be approximately 10W lower. The piston position was easily controlled by controlling the back end pressure and no instabilities were encountered.

The first time the unit was installed in the test rig and run as a refrigerator it lifted 200W from -23 C to ambient at a COP of 1.5. This is a promising result since the cycle had not been optimized yet. The delivery of the compressor was easily modulated by adjusting its supply voltage. Very little vibration was transferred through the mounting springs to ground. Noise levels are similar to those for a conventional compressor - the dominant noise is that created by the inlet valve.

Expected performance

The experience gained thus far, combined with the results of the computer simulation indicated that a COP of 2.0 or better is achievable. With suitable control and optimization it is expected that this efficient operation can be maintained over a wide range of heat lifts and ambient conditions.

CONCLUSIONS

The members of the design team believe that this compressor will revolutionize domestic refrigeration. It promises to be 30 percent more efficient than existing units and has the added advantages of allowing easy, continuous modulation of heat lift and lubricant free operation. It is eminently suitable for refrigeration cycles using HFC134a or other ozone friendly refrigerants.

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